Central Institute for Scientific and Technical
Information and Technical and Economic
Research on Chemical and Petroleum
Machine Building

KhM-3
Oil Industry
Machine Building

Information Summary

IMPROVING THE EFFECTIVENESS OF DRILLING TOOLS

TSINTIKHIMNEFTEMASH
MOSCOW, 1991

CENTRAL INSTITUTE FOR SCIENTIFIC AND TECHNICAL INFORMATION
AND TECHNICAL AND ECONOMIC RESEARCH ON CHEMICAL
AND PETROLEUM MACHINE BUILDING

Information Summary

PETROLEUM INDUSTRY
MACHINE BUILDING

Series KhM-3

**B. L. Steklyanov, A. V. Torgashov, A. A. Loginov,
Yu. N. Sokolov and K. G. Valiyeva**

IMPROVING THE EFFECTIVENESS OF DRILLING TOOLS

INTRODUCTION

An optimization problem in the field of drilling tools generally should be presented as a problem of optimizing the process of rock breaking in drilling. This interpretation has made it possible to solve these problems two ways: by minimizing the energy cost of breaking rock within a specific cylindrical space congruent with the bottom hole, and minimizing wear of the bit equipment, and by increasing the intensity of the effect on the bottom hole. Both methods are special cases of the general solution of the optimization problem. However, the solution of the optimization problem by the first method, which involves the theory of rock breaking, is accessible to a limited number of specialists.

The second method is the most acceptable and effective approach. This summary presents the bases for this method within the scope of design restrictions, specific examples of its practical use in the design of a new, more effective rock-breaking roller cone bit, and prospects for its development and use in the domestic bit industry.

The methodology for solving optimization problems has been developed based on a determinate analytical model of drilling bit operation constructed and perfected at the Central Asian Research Institute of the Gas Industry (SredAzNIIGiprogaz). The model makes it possible to compute kinetic criteria for assessing the serviceability of drilling bits as a function of their geometric parameters. This capability is an objective prerequisite for the statement and guaranteed solution of a particular optimization problem. The point is the particular solutions that make it possible to achieve a significant saving.

One can conclude that the potential of the method for improving the effectiveness of domestic roller cone bits by improving their geometry has not been nearly exhausted. Therefore, mathematical modeling as a basis for solving optimization problems in the field of rock-breaking drilling tools is timely and promising.

MECHANICS OF ROLLER CONE BITS

In investigation of the mechanics of rock breaking under the effect of bit cones rolling on the bottom hole, questions of qualitative and quantitative impact arise, along with questions of the reaction of the rock to the roller teeth by rows in regard to strength and abrasive aspects. It is these questions, in particular, which necessitate finding objective criteria to assess the serviceability of a rock-breaking tool. Without knowledge of the actual kinematics of the roller equipment, it is difficult to visualize the overall picture of the mechanism of interaction of the roller teeth with the bottom hole, much less construct an analytical structure of their kinetic characteristics that objectively reflects this mechanism and depends upon it.

The kinematics of the cones and, consequently, their cutters generally depends on the geometric parameters of the drilling tool and the drilling conditions [8]. The kinematic characteristics can be expressed in relative units as a function of the bit geometry [10]. This simplifies the problem considerably and makes it possible, as a result, to construct (compute) and analyze relative kinetic criteria for assessing the serviceability of drilling bits and to use them in solving optimization problems in the process of improving the kinetics of cone cutters. For this purpose, it is necessary to know the physical nature of the criteria. Special attention is devoted to this issue in the summary, since ignoring any factor of the mechanism of interaction of the cone fittings can result in design calculation errors in the process of designing drilling bits.

A determinate analytical model of the operation of drilling bits on a deformable bottom hole has been described in previous studies [9, 16]. Therefore, this summary considers only the basic issues of the construction of the model.

1. The three-dimensional trajectory of movement of a cone tooth is well described by parametric equations in a Cartesian coordinate system [11],

$$\left.\begin{aligned} X &= x(n,G,i) \\ Y &= y(n,G,i) \\ Z &= z(n,G,i) \end{aligned}\right] \qquad (1)$$

where n is the rotation frequency of the bit; G is the combination of geometric parameters; i is the gear ratio of the cone.

According to the quality of interaction of the cone teeth with the bottom hole, these trajectories may have 27 different forms, which depend upon the incline angle of the cone axis, the horizontal offset of the rotation axes of the cones, and the ratio of the angular velocity of the cones and that of the bit [9]. This relationship is observed in the lower regions of the trajectories. The trajectories of teeth of peripheral rows that gauge the hole walls are no exception. One must keep in mind, however, that the cone teeth cone teeth are in contact with the rock in considerably larger sections.

However, this pertains to the quantitative evaluation of the mechanism in question, i.e., to integration of the paths of contact of the teeth of the respective rows.

2. The path of contact in general form is defined by the formula

$$S_j = \frac{\pi n}{30} \int_{\Psi_0}^{\Psi_1} v_j(\Psi) d\Psi_1 , \qquad (2)$$

where Ψ_0 and Ψ_1 are the integration limits, in degrees; $v_j(\Psi)$ is the linear velocity of movement of a cone tooth, in mm/sec.

The analytical structure of the function $v_j = v(\Psi)$ is quite complex for performing a qualitative analysis. However, the cone rows that have positive or negative slippage are established in the process of calculations. The clear rolling row is also known in this process, so that the qualitative side of this function is clearly known in each individual case. Hence for the purpose of introducing great clarity to the qualitative and quantitative aspects of the value of S_j, the whole range of roller cone bits is divided into five classes, each of which has its own formation and integration features [9].

In addition, the function $v_j = v(\Psi)$ depends on the gear ratio of the roller cones and can be determined experimentally or computed with a sufficient degree of accuracy.

3. The gear ratio of the cones is computed by the method of sequential approximations based on the principle of lowest power consumption [3]. In general, this method can be expressed in the following form [10]:

$$\sum_{j=i}^{m} A_j = \sum_{j=m+i}^{n} A_j \ , \tag{3}$$

where A_j is the work of the cone teeth in the j row, in J; n is the number of rows; m is the number of rows from the periphery to the clear rolling row.

The essence of this method is that with the true value for the gear ratio of the cones, the work from friction forces of the teeth with positive slippage is equal to the work of the rows with negative slippage.

The question of the gear ratio of the cones became critical as soon as the cone geometry began to differ from the form of a perfect cone. Currently there is no drilling bit with roller cones with a perfect cone shape. Hence a procedure for computing the gear ratios of the cones as a function of their geometry was necessary. This is logical, since a change in the form of the roller cones led to indeterminate values of their gear ratios. The calculation by this procedure is quite extensive, but it presents no special difficulties with the use of computer equipment. The gear ratios of the cones can be determined with an accuracy of up to 5 – 10% without special work. It is recommended that data accumulated in the course of the research be used to define approximately the annular zones of the bottom hole with different slippage depending upon the gear ratio of the cones.

Hence for tri-cone bits with horizontally offset cone rotation axes and without offset axes, 1.3 – 1.5 may be considered the limits of variation of the gear ratio. Disk bits, as a rule, have no more than 2-3 gauging sides, and one of them is the leading row, i.e., the clear rolling row. In this case, the gear ratio

$$i = \frac{R}{\tau}, \tag{4}$$

where r is the radius of one of the gauging sides, in mm; R is the radius of the circle within which the gauging side selected for the calculation rolls.

For single-cone bits, as studies have demonstrated, with an angle α of 0 - 90° between the cone and bit axes and a relatively even distribution of cutters on cones of any diameter, the gear ratios are as follows:

α°	10	20	30	40	50	60	70	80
i	0.9	0.78	0.64	0.5	0.4	0.32	0.22	0.1

For the purpose of qualitative evaluation of the mechanism of interaction of the cone fittings of drilling heads, one can use the result of study [7] in the first approximation, adopting as the clear rolling row the row which is 1/3 of the length of the generator of the cone from the peripheral row.

4. It is conventional to consider the specific volume and contact work of rock breaking as the criteria for quantitative evaluation of the effectiveness of the work of drilling bits [2]. In light of the fact that the mathematical model of the work of drilling bits includes in explicit form only their geometric parameters, while the bottom hole (rock) conditions and energy parameters are included in the model indirectly as the assigned value of sinking of the cone teeth into the rock, the following evaluation criteria have been adopted: the relative specific contact work A'_j and volume work A''_k of rock breaking [16], N·m/m^3.

$$A'_j = S_j F_j \ ,$$
$$A''_k = \frac{\sum_{[j=1]}^{n} S_{jk} F_j z_j d_j}{V_k} \qquad (5)$$

where F_j is the force of resistance to the movement of teeth of the j row in contact with the rock, N; z_j is the number of identical teeth in the j row of the cone; d_j is the number of conventional rows of unit length making up the actual row; V_k is the volume of rock to be broken in the k ring of the bottom hole, m^3.

The physical nature of these criteria is clear from analysis of the formula (5).

The value A'_j is an analog of the relative intensity of abrasive wear of the fittings of adjacent rows and rows of adjacent cones. The same kind of analogy is seen in analysis of basic and new designs for drilling bits of the same standard size. The relationship is also characteristic of relative endurance criteria of bits, on the condition that the relative endurance is not limited by the endurance of bearings of adjacent cones.

The value A''_k is an analog of the relative intensity of rock breaking for adjacent bottom hole rings and, consequently, the sinking of the teeth, or the mechanical rate of drilling. This criterion obviously is also an analog of the relative stress level and endurance of the fittings and bearings of adjacent cones.

With the bit design criteria in tabular or graphic form, it is possible to perform an objective analysis of the serviceability of the bit. From such an analysis, the weakest points of the drilling bit become obvious based on the kinetics of the fittings. The extreme values of A'_j and A''_k, namely A'_{max} and A'_{min}, are of decisive importance in this process. Interpreting what has been said, one can state that this kind of analysis reveals with sufficient accuracy the rows that are most susceptible to abrasive wear (the criterion A'_{max}), as well as the rows subject to heavy breaking, and the bearing, loaded with a probability of failure ahead of it (the criterion A''_{min}).

It is worth noting that in comparative analysis of the serviceability of the fittings of drilling bits, it is necessary to exclude certain factors capable of causing negative phenomena and

to reduce their kinetic potential; i.e., it is necessary to make the appropriate adjustments to the following basic factors based on experience.

Bottom hole form. A comparative analysis of the bottom hole form is conducted from the point of view of the rock's resistance level to breaking. It should not be greater than the basic bit design for the entire bottom hole. If it is impossible to fulfill this condition, the introduction of compensating factors must be justified scientifically.

In regard to **strength of components**, the bit to be designed should at least be no worse than the basic model. In calculations, it is desirable to start with the condition of at least a threefold strength margin.

Gear ratio of cones. The need for a correction lies in the fact that each of the adjacent cones has its own gear ratio. Hence the wear of the fittings of the cones and the bearing surfaces, as well as the intensity of the effect on the rock, must be taken into account per unit of time, in light of the varying rotation frequency of the cones.

Number of teeth in the rows. The need for such a correction is obvious, because, for example, the more teeth are present in one of the equally loaded rows of adjacent cones, according to the criterion of equality of the sinking of the teeth with time, the more loading pulses will be transmitted to it, and, consequently, the faster the moment of critical bearing wear and fatigue failure of the teeth may arrive.

Wear of fittings and bearings with time. During the working process, due to abrasive wear, the bit geometry is altered. Therefore, the kinetic criteria, as functions of the geometric parameters of the bit, also change. Consequently, the direction of the change and the limits within which it occurs as compared to the corresponding parameters of the basic design must be known and taken into account.

Cost criterion. Such an assessment must be based on the actual hole depth at which the bit will work and the mechanical rate. This will make it possible to assess and predict the cost of operations more accurately in the conceptual design phase.

This is the system for solving the problem of optimizing a roller cone rock-breaking tool.

The limitations imposed and the verification and correction of the system obviously make it logically complete and improve its quality. On the other hand, these factors result in significant complications in regard to scale and logic factors.

Kinetics of Paired Cone Teeth

Cases which may be encountered in the process of designing drilling tools (bits) are considered below. The bit components are considered in a state of dynamic equilibrium, on which the thoroughness with which their potential is utilized depends to a considerable degree.

One of the most important problems in the design of fittings of roller cone bits is thorough drilling with coverage of annular bottom hole sections with bit nose rows. In this process, blocks, collars and rods must not be allowed to form. These phenomena can be eliminated by enlarging the contact surfaces of the respective bit rows, which increases the coverage factor for the annular bottom hole sections to be drilled. This is achieved by increasing the number of teeth or by introducing shaped teeth, such as Z, Γ and T shapes. This method is effective when it is not possible to add more teeth to the cone rows, or with a comparatively high coverage factor. Bits with paired teeth are of great interest. The 3S-K three-roller bit of the Tsukomoto Seiki Company (Japan), with a diameter of 295.28 mm, is one such bit. A twin tooth,

the spacing of which is half that of the adjacent teeth, is found in the second row from the edge of the second cone of this bit.

In the use of these bits in a field of the "Uzbekburgaz" Production Association, shear fracture of the leading tooth of the pair was often observed. Special studies were performed to discover the cause of this phenomenon. The kinematic scheme of the coverage of the annular bottom hole by a row with paired teeth and the establishment of the causes of shear fracture of the leading tooth are of interest.

We shall examine the mechanism of the interaction of the teeth of the row in question with the rock. It was found, after calculation of the kinetic characteristics of the operation of this bit by the SredAzNIIGiprogaz method, that the row of the second cone with paired teeth does not operate in a clear rolling region.

We shall proceed with calculations and analysis based on the geometric parameters of the row and the cone characteristic of this bit: row radius $r = 81$ mm; radius of annular bottom hole to be drilled $R = 101.6$ mm; number of teeth $z = 11$; gear ratio $i = 1.31$; spacing of paired teeth $l = 26.7$ mm.

Since the gear ratio of the second cone $i_0 = 1.31$, and the ratio of the average radius of the annular bottom whole covered by this row to the row radius $i = 1.25$, it follows from the inequality $i < i_0$ that the row in question operates in a region of negative slippage.

The reaming length of the row, taking into account the gear ratio (for a single revolution of the bit), $S_1 = 2\pi ri = 666.7$ mm. In this case, the average circumference of the annular bottom hole in question $S_2 = 2\pi R = 638.4$ mm. Consequently, in a single revolution of the bit, a cone row with a reaming length of 666.7 mm processes a bottom hole ring with a circumference of 638.4 mm.

The difference $S_1 - S_2 = 28.3$ mm is the amount of slippage for 14 teeth of the row which enter into contact with the rock in one revolution of the bit. Consequently, the amount of slippage of each of the teeth averages ≈2 mm. It follows that the geometric spacing of the teeth is less than the kinematic one.

A diagram of the interaction between the row of the second cone with paired teeth and the bottom hole has been constructed based on calculations. The teeth work almost in each other's tracks, leaving unprocessed sections (Fig. 1, shaded) the length of which is equal to the distance between adjacent teeth, which indicates that the fittings have been designed without taking slippage into account.

The mechanism of the interaction of leading and dependent teeth of the pair in question can be represented in the following form (Fig. 2). Since the paired teeth go into the same hollows that unpaired teeth have dug before only every few revolutions, their profile will be different. Moreover, as already mentioned, the spacing of paired teeth is larger than the spacing of the hollows (the kinematic spacing of the teeth).

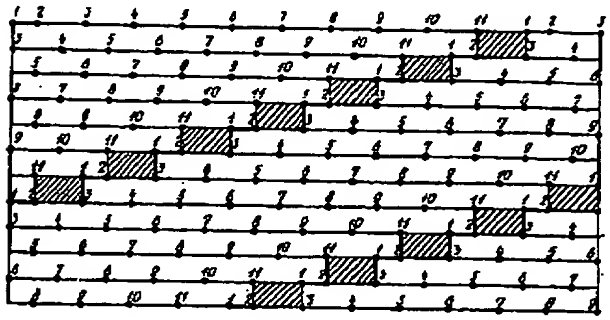

Fig. 1. Diagram of the coverage of an annular bottom hole by a row with paired teeth.

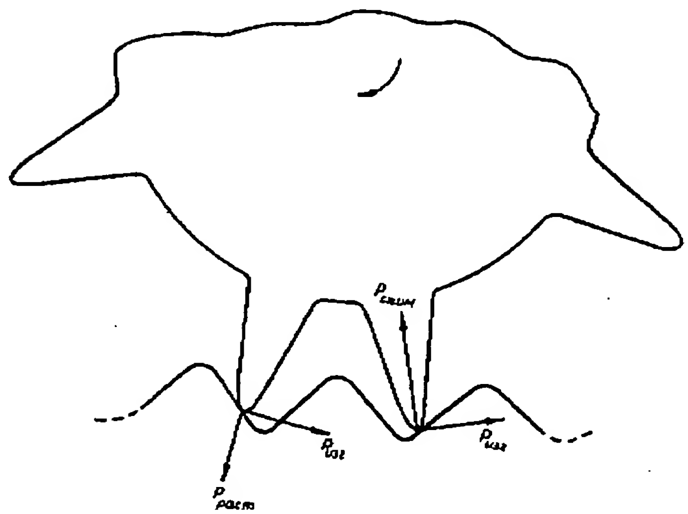

Fig. 2. Diagram of the interaction of paired teeth with furrows on the bottom hole.

Since the row operated in a negative slippage mode (side cutting mode), one can draw the following conclusions concerning the loading of these teeth: the leading tooth will experience [words cut off in source document] . . . ; the probability of [words cut off] will always be greater as a result of the fact that it undergoes bending and stretching simultaneously.

Hence studies have demonstrated that the shear fracture of the leading tooth of this bit is not random, and the bits are intended for drilling out rock of medium hardness. It is also obvious that including paired teeth in the bit design can be justified, if the mechanism of interaction of the cone teeth with the bottom hole is taken into account objectively in this process. In the case in question, the introduction of a paired tooth is not justified, since shear fracture of the leading tooth can occur during the very first minutes of operation of the bit; i.e., the bit is dynamically unbalanced.

Effect of Power from Forces of Resistance to the Movement of Solids on Their Dynamic Balance

The phenomena that arise in drilling holes and the vibration of rotating shafts with disks at critical revolutions [4, 5, 12] produce a graphic representation of dynamic instability of solids in rotating movement. However, such factors as an (n+1)-sided hole cross section with n drill bit edges and damping of oscillations of rotating shafts with disks with an increase in the rotation speed still remain unexplained.

In the study of these phenomena, insufficient attention is being devoted to the forces of resistance to movement – friction of the drill and disks against the respective outside objects.

We shall perform a preliminary study of the effect of the arguments of analytical structures of the powers of rotating bodies on the magnitude of these powers.

Assume that a body in the form of a disk rotates in a plane *P* around two parallel axes with a different predefined eccentricity ε and gear ratio I (Fig. 3) under the effect of external forces overcoming friction.

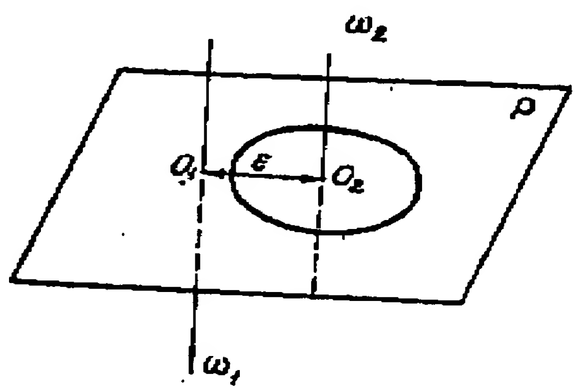

Fig. 3. Diagram of the movement of a disk around two parallel axes.

We assume that the angular velocity ω is constant. This assumption simplifies the calculations and, at the same time, makes it possible to trace the physics of the topic of the study more deeply. In this process, in particular, the power from friction forces in the rotation of the disk only around its own axis N_0 takes on a totally specific value, the comparison of which to the power spent by the disk in rotation in different modes around two parallel axes is fully justified.

Assuming the friction coefficient and the axial force applied to the disk to be constant, we construct the following function (Fig. 4),

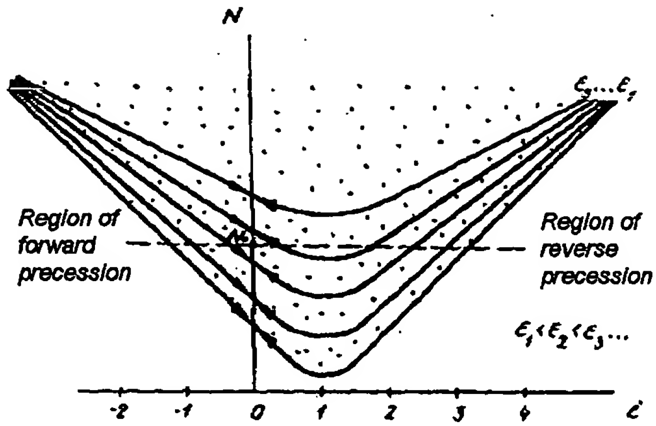

Fig. 4. Power expenditure as a function of forces of resistance to movement in rotating motion.

$$N = N(\varepsilon, i), \tag{6}$$

where N is the power created by forces of friction of the disk with a surface, kgf·m/sec; ω_1 is the angular velocity of rotation of the disk around the first axis; and ω_2 is the angular velocity of rotation of the disk around the second axis, sec^{-1}.

The distinguishing properties of the function are as follows:

1. Does not depend on the resistance medium or the form of the body;

2. Minimum is at the point i = 1 (rotation pair);

3. Minimum is in direct proportion to the magnitude of eccentricity;

4. Growth of the function is symmetrical in relation to the point i = 1;

5. Function has bifurcation points at i = 0 and $\varepsilon = 0$;

6. At the bifurcation points characterizing the emergence of the disk in a mode of rotation around a single axis, the function shows an abrupt increase.

The function obtained for the general case can be represented as a function of power expenditure. The power spent to overcome forces of resistance to movement is expressed in the form of a nonlinear function of the gear ratio, with a minimum directly proportionate to eccentricity in rotation pair modes.

The jump increase in power at i = 0 and $\varepsilon = 0$ is conditioned by the fact that the disk enters a mode of rotation around a single axis in which the power N_0 is expended.

Taking into account the properties *3* and *6* of the function (6), one can easily explain the capability of rotation of a bit around two parallel axes and an (n+1)-sided hole cross section. On the one hand, according to the principle of least power expenditure and property *6*, the bit actually goes into a mode of rotation around two parallel axes, since there are many modes of rotation around two parallel axes below the level of N_0, i.e., with less power expenditure. On the other hand, according to property *2*, eccentricity should be at a minimum in this case, which is provided when the faces are equal only in the case of rolling of an n-faced polyhedron in an (n+1)-faced polyhedron. Hence the factors that destabilize the dynamic equilibrium in this case are frictional forces, and the hole walls only streamline the vibration process of the drill. It is extremely difficult to avoid this effect of the pattern obtained in drilling holes, which in practice necessitates reaming to obtain a round hole cross section.

Hence the regulated movement of drill bit edges is determined by the rotation mode with the least power expenditure within the constraints imposed by deepening. If we assume rotation of the drill in contact without deepening, the process of the rotation mode with the least power expenditure will be disordered, which is observed at the start of drilling. This search is performed in a region of negative precession, which conforms well to formula (6).

This is a starting point for discovering the nature of critical numbers of revolutions of rotating shafts with disks. For fulfillment of the requirement of minimum power consumption, according to property *6*, the disk on a rapidly rotating shaft in contact with some medium will actually try to enter a mode of rotation around two parallel axes. However, this must occur at a low friction coefficient and quite a high number of revolutions, where the power created by internal forces will already be insufficient to overcome flexure. Since the power from forces of friction of the disk against the medium in rotation around a single axis increases in proportion to the number of revolutions, this state of the disk is possible at constant power. It is important to note at this point that with an eccentric position of the disk due to bending of the shaft, as in the previous case, the process of searching for a mode of rotation around two axes with the least power expenditure will be initiated as this eccentricity tends toward zero (property *3*). However, this reverse process (straightening of the shaft) will now be accelerated, since it is promoted by the elasticity of the bent shaft and property *3* of function (6). This explains the presence of sustained vibrations of the shaft in rotation at critical revolutions.

Consequently, the nature of the vibrations of a drill and of a rapidly rotating shaft with a disk is the same. It is included in the properties of the pattern obtained above for power expenditure [see formula (6)]. The difference here lies only in the fact that the vibrations of a drill and a disk arise at different rotation frequencies, which is quite natural.

We shall attempt now to investigate the cause of damping of vibrations with a further increase in the rotational frequency of shafts with disks. It is natural to suggest that gyroscopic forces begin to have a greater effect here. For the sake of clarity, we shall present the summation of powers from the forces of this triad in graphic form (Fig. 5), i.e.,

$$N_p = N_0 + N_1 + N_2 = N(F, G, J, n), \tag{7}$$

where N_p is the power of the dynamic system, J; $N_1(G, n)$ is the power from internal forces of the shaft, J; $N_0(F, n)$ is the power from forces of friction of the disk against the medium in rotation of the shaft around a single axis, J; $N_2(I, n)$ is the power from gyroscopic forces, J.

The nature of the forces of powers N_1 and N_2 obviously promotes stabilization of the shaft in relation to its axis, while the force of power N_0 promotes destabilization, which has been taken into account by the signs in plotting the graph. The graph of function (7) clearly illustrates the nature of critical numbers of revolutions of rotating shafts with disks.

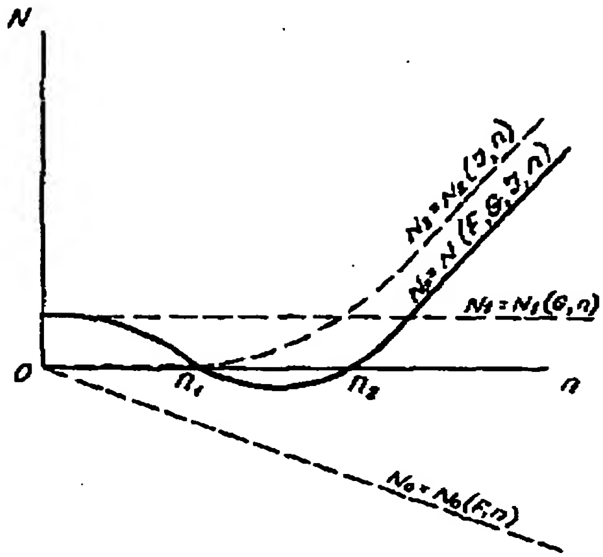

Fig. 5. Curve of the power of a rotating shaft with disk.

However, the forces of friction of bodies against a medium in dynamic systems have a dual nature. In the rotation of a body around a single axis, these forces, like centrifugal forces, try to draw it into a mode of rotation around two axes, but as soon as the body enters a mode of rotation around two parallel axes the same forces of friction, in reducing ε (property *3*), try to draw the body into a mode of rotational around a single axis. Therefore, the assertions of a one-way effect of friction forces on the equilibrium of dynamic systems were mistaken, which disrupted the objective representation of the physics of their vibration processes.

It is also known that the transition of bodies to different rotation modes is associated with a sharp increase in power expenditure. This causes oscillations of all kinds and vibrations of a dynamic system with friction pairs. Ordering or stabilizing the oscillations of such a system is possible only by finding a rational combination of parameters of function (7).

The function of N_p (see Fig. 5) with the appropriate geometric and physicomechanical parameters of the dynamic system will be valid in the range $n_1 \leq n_{cr} \leq n_2$. We shall now consider function (7) in regard to the bottom-hole operation of a bit. It is important here to note two cases.

1. Since the operation of the bit proceeds under significant axial loads, the function $N_0 = N_0(F, n)$ forms a large angle with the *0n* axis.

2. The rigidity of the drilling head is near zero; i.e., the function will be close to the *0n* axis.

Based on these prerequisites, one can visualize a drilling column as an unstable system, because the real rotation frequency is always in the critical range. The same entire pattern of power expenditure is the main cause of dynamic instability of the drilling bit at the bottom hole.

Dynamic Equilibrium of a Spherical Cutter

A single-cutter bit with a cutter of spherical form has a tendency to deviate from its axis. However, it is wrong to think in this case that the reason for this is the fact that the bit and cutter axes do not coincide.

We shall assume that the cutter axis coincides with the bit axis. In this case, the bit will be of the cutting-grinding type. The conclusions presented above confirm that such a bit will deviate from its axis. It will constantly strive for a minimum of power expenditure in a mode of rotation around two parallel axes. This explains the dynamic nonequilibrium of spherical cutters, since the presence of an additional rotation axis defines an additional degree of freedom, which allows the cutters to go into the mode of a rotation pair with minimally varying eccentricity.

However, one cannot entirely exclude the role of asymmetry in the design of single-cutter bits. Drilling rock with single-cutter bits is characterized by the fact that the peripheral areas of the bottom hole are broken with greater intensity that the central area. Otherwise, there would be self-centering of the bit in relation to its axis. Sticking of the spherical cutter on the rock in the central area of the bottom hole is observed in this case. It is just this circumstance which provokes the bit to deviate practically unobstructed from its axis. The asymmetry of rock breaking is only a limiting condition and not the cause of radial movements of the bit in relation to the bottom hole.

It is known that the smaller the incline angle of the teeth row of a spherical cutter α in relation to the hole cross section plane, the smaller the length of contact of the teeth with the bottom hole in one revolution of the cutter, and the smaller the difference between the lengths of contact paths of the teeth of adjacent rows (Fig. 6).

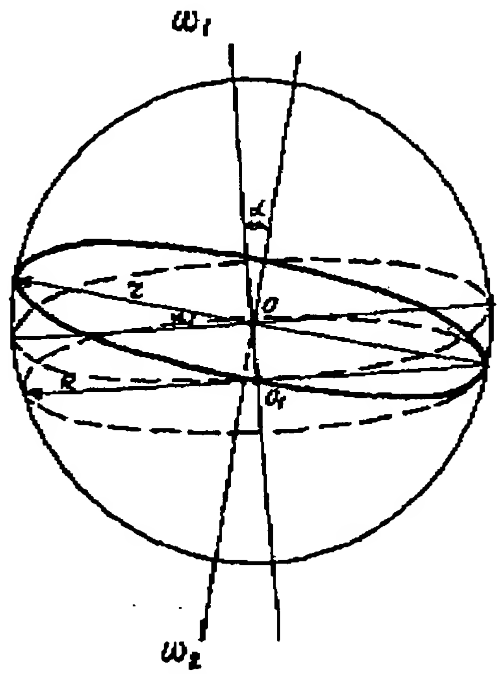

Fig. 6. Diagram of the teeth row position of a spherical cutter.

Obviously, as $\alpha \to 0$, $r \to R$, (8)

i.e., $i = \frac{R}{r} \to t$. $i = \frac{R}{r} \to t$

Consequently, as $\alpha \rightarrow 0$, the dynamic system in the form of a single-cutter bit tends toward a mode of rotation around two parallel axes, or, more precisely, toward the mode of a rotational pair, in which the lengths of the paths of contact of the points are equal and are equal to the length of a circle with a radius equal to the eccentricity, i.e., if the trajectories of the teeth of these rows at sufficiently small angles α are close to the length of circles of equal radii. It is important to take this condition into consideration, not only in examining the dynamics of single-cutter bits, since the power expenditure (6) are regular for all dynamic systems with forces of resistance to the movement of their elements. In this case, the forces in question are friction forces. Consequently, the dynamic equilibrium of drilling bits of any design must be considered in the light of the proposed formula for power expenditure. This is important in the design of new rock-breaking drilling tools.

It is clear from the information presented that self-centering of drilling bits in relation to their axes is impossible. The bits always tend to go into a mode of rotation around two parallel axes. This process obviously is cyclical and ordered.

The stabilization of the rock-breaking tool is important in this regard and can be performed in two ways: due to dynamic congruence of the surfaces of the working bit and the bottom hole, and with a stabilizer.

It must be taken into account, however, that in the design of stabilizers, paradoxical phenomena which are similar at first glance may be encountered.

Dynamic Equilibrium of Stabilizers

We shall examine a stabilizer design based on the principle of positioning the rollers in three planes at some distance apart in relation to the housing axis [6]. The rollers have a spiral offset of 120° in relation to each other (Fig. 7).

It has been proven experimentally that stabilizers of this type have a tendency toward shifting their axes. It has been noted that their minimum diameter is less than the diameter of the circle of the cross section of the hole, against the walls of which the rollers would have to rest in stabilizing the bit on the bottom hole. If the stabilizer rollers are inscribed in a circle with a diameter equal to the bit diameter, there will be no stabilization effect.

One can see from Fig. 7 that the diameter of the channel through which a lens with a height equal to the length of the roller generator will pass will be significantly less than the nominal diameter of the hole. Therefore, it is necessary to know the thickness of the lens or the lengths of the stabilizer roller generators at which the channel diameter will be greater than the nominal diameter of the hole. Thus a triangle *ABC*, in rotating around an axis BB'_1, can pass through an opening with a diameter $BE = BB'_1$, and

$$BE = \sqrt{(R_{pac}\sqrt{3})^2 - \left(\frac{R_{pac}\sqrt{3}}{2}\right)^2} = \frac{3}{2}R_{pac} \quad . \tag{9}$$

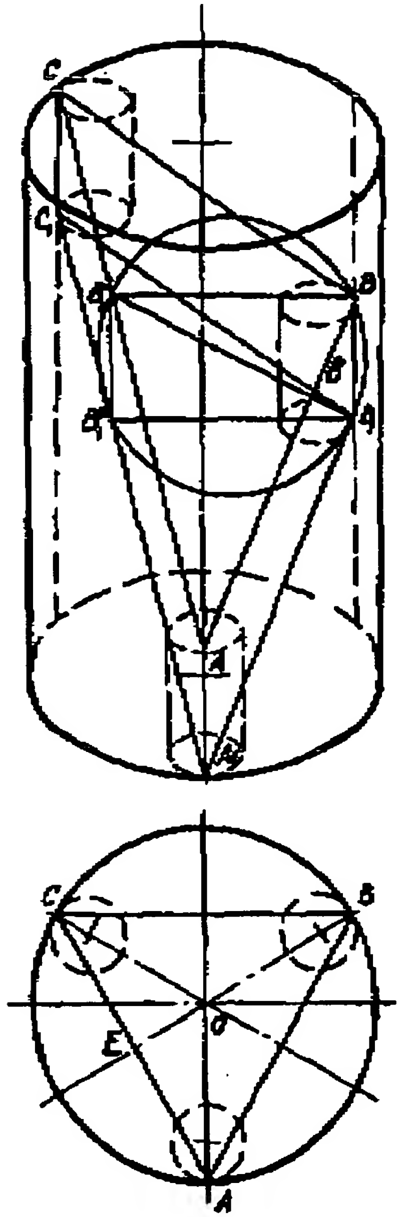

Fig. 7. Diagram of three-cutter stabilizer.

We find the channel diameter $D_0 = B'B_1$ from the triangle $BB_1B'_1$, where $BB_1 = \sigma$ is the length of roller generator,

$$D_0 = \sqrt{\left(\frac{3}{2}R_{pac}\right)^2 + \sigma^2} \ , \qquad (10)$$

from which

$$\sigma = \sqrt{D_0^2 - \left(\frac{3}{2}R_{pac}\right)^2} \ . \qquad (11)$$

Considering that $D_o = 2R_{pac}$, we find

$$\sigma = \sqrt{\left(2R_{pac}\right)^2 - \left(\frac{3}{2}R_{pac}\right)^2} \ ,$$

or

$$\sigma = \frac{R_{pac}}{2}\sqrt{7} \ . \qquad (12)$$

Consequently, for a three-roller stabilizer to perform its function of stabilization of a bit at the bottom hole, the length of the roller generator must be greater than the nominal radius of the hole by the length $\sqrt{7}/2$, i.e., the following condition must be fulfilled:

$$\sigma > \frac{\sqrt{7}R_{pac}}{2} \ . \qquad (13)$$

Ignoring the condition (13) leads to dynamic destabilization of the bit at the bottom hole. This must be taken into account in designing stabilizing elements for drilling tools.

One must keep in mind that the fulfillment of the condition (13) is not the only solution of the problem in question. It can be solved by including a fourth roller in the stabilizer design or, for example, by lengthening one or two rollers. Only with objective assessment of the dynamic equilibrium of the drilling bit at the bottom hole can one analyze with sufficient precision the mechanism of the interaction of the cutters of roller cone bits and solve the optimization problem with improvement of the bit designs.

IMPROVEMENT OF THE DESIGNS OF MASS-PRODUCED ROLLER CONE DRILLING TOOLS

The improper statement of optimization problems in the field of rock-breaking drilling tools design is conditioned by multiple-criterion assessment of their serviceability. The specific nature of the solution of these problems is also conditioned by the extremely complex functional

dependence of the criteria for assessment of the effectiveness of drilling bits on the system of bit – rock – energy [1, 13, 14]. Nevertheless, the statement and effective solution of these problems in their technical and technological aspects is possible. We shall consider methods that make it possible to accomplish the purposeful and sufficiently expeditious determination of solutions, the basis of which is correspondence of the local changes in the parameters of drilling bit components to extreme values of the kinetic criteria [see formula (5)].

The basic causes of failure of a drilling bit are abrasive wear of the teeth of the cone teeth rows, shearing of the teeth, and wear of the bearing surfaces. These causes are the result of an uneven distribution of the specific contact and volume breaking work, respectively, on adjacent rows and the annular bottom hole, as well as fully defined and stable coordinates of their extreme values, which significantly facilities the statement and solution of the optimization problems.

Particular methods have been generated in the processes of solving optimization design problems.

1. Covering annular bottom holes with a large contact surface area with a steady cone rotation mode. The method is effective with an insufficient value of A''_{kl}, which under certain conditions affects the mechanical drilling rate, the bearing overload of the respective cone, and the strength of the teeth.

2. Changing the cone rotation mode for covering the annular bottom hole with slipping rows. The method is effective at a low value of A''_{kl}, as well as under conditions in which rock breaking is difficult.

3. Shifting the values of $A''_{k\,min}$ into a zone with easier rock breaking conditions. This method is universal, other conditions being equal.

4. Minimizing (to the extent this is permissible) the values of $A'_{j\,max}$ in designing roller cone bits with horizontally offset cone rotation axes. The method is effective in the design of roller cone bits for drilling in abrasive rock. This feature is characteristic of a heel row. It can be varied by changing the amount of the horizontal offset of the cone rotation axis and the radii of the gauge rows. The method of doubling gauge rows is also effective in this case.

5. Evenly distributing the load among the sections of the bit. The method is effective in leading wear of one of the cone bearings. The effect is achieved by balancing the minimum value of A''_k on all the cones and will be even higher with an increase in $A''_{k\,min}$.

6. Balancing A'_j among the rows in light of equal-strength conditions at the bottom hole. The method is effective for bits that form a spherical bottom hole, such as single-cutter bits.

The system or strategy for solving the optimization problem in the field of rock-breaking drilling tools is represented in general form in Table 1 [16].

Table 1

Solution of a reverse design problem based on changing		Cause of bit failure	Solution of a reverse technological problem based on changing		
material	geometry		geometry		parameters of drilling mode
Strengthen the row with more durable material	Change the bit geometry so that $A'_{j\,maxN}<A'_{j\,maxO}$ at $A'_{j\,minN}>A'_{j\,minO}$	Wear of row teeth (defined by value of $A'_{j\,max}$)	Select a bit with $A'_{j\,maxN}<A'_{j\,maxO}$ at $A''_{j\,minN}\geq A''_{j\,minO}$	Select a bit in which the row material is more durable	$P_N>P_O$ $n_N\leq n_O$
Strengthen the row with stronger material	Change the bit geometry so that $A''_{j\,minN}>A''_{j\,minO}$ at $A'_{j\,maxN}\leq A'_{j\,maxO}$	Shearing of row teeth (defined by value of $A''_{j\,min}$)	Select a bit with $A''_{j\,minN}>A''_{j\,minO}$ at $A'_{j\,maxN}\leq A'_{j\,maxO}$	Select a bit in which a row of stronger material [illegible]	$P_N<P_O$ $n_N\leq n_O$
Strengthen the	Change the bit	Wear of bearing	Select a bit with	Select a bit with	$P_N\geq P_O$

bearing with more durable material	geometry so that $A''_{j\,minN}>A''_{j\,minO}$ or $A''_{j\,minI} \approx A''_{j\,minII} \approx A''_{j\,minIII}$	(defined from $A''_{j\,min}$ of adjacent cones)	$A''_{j\,minN}>A''_{j\,minO}$ or $A''_{j\,minI} \approx A''_{j\,minII} \approx A''_{j\,minIII}$	a more durable bearing or a sealed bearing	$N_N<n_O$

N o t e : The indexes O and N refer to old (base) and new bits, respectively.

As one can see from the data of Table 1, the extreme values of the kinetic parameters that cause failure of a bit are also valid for the technological aspect – selection and effective optimization of drilling bits under given geological engineering conditions. This is quite natural, since the kinetic criteria are functions of the system of bit – rock – energy. Consequently, there should be an inverse relationship, or the general solution of an inverse problem represented in the indicated system (see Table 1), which is the same thing.

Without going into a detailed analysis of the system for general solution of the inverse problem, we shall only give a few specific examples of the solution in the form of recommendations for improvement of rock-breaking drilling tools and their effective optimization.

Single-Roller Drilling Bit I 161SZ-N

A single-roller bit fails, as a rule, after the potential capabilities of the cutters have been exhausted. The wear of teeth on the rows of a spherical cutter is local; i.e., leading wear of the cutters occurs on one or several row points.

For scientifically substantiated explanation of the causes of failure of drilling bits, SredAzNIIGiprogaz constructed a model of their operation on a deformable bottom hole, which makes it possible to compute the relative kinetic criteria A' and A'' quickly by computer and to perform a fast analysis of the serviceability of the drilling bits. The criterion A' – the specific contact work of rock breaking – defines the relative intensity of the wear of cutters by rows of the roller. The criterion A'' – the specific volume work of breaking – defines the relative intensity of rock breaking in round strips of the bottom hole.

Based on the value of A'_{max}, one can predict the row most susceptible to wear, i.e., the row whose serviceability (life expectancy) determines the durability of the bit. The degree of rational use of hard-alloy cutters is determined based on the ratio of values A'_{min}/A'_{max}. To compare the durability of base and new bit designs, it is necessary to use the values of $A'_{max}i$, i.e., the maximum value of the intensity of wear of roller cutters per unit of time (per revolution of the bit).

Table 2

Geometric parameters					Kinetic characteristics		
Conventional row number J	Row radius Rj	Bottom hole radius Ro	Number of teeth in row	Row width D	Tooth speed v	Specific contact work A'	Specific volume work A''
1	77.85	77.67	9	1	57.49	146.08	219.3
2	80.50	69.96	18	1	41.41	195.81	87.2
3	78.10	57.89	9	1	22.77	242.11	42.7
4	70.14	41.00	9	1	1.29	356.05	21.1
5	47.34	21.41	9	1	-20.04	402.10	15.5

6	42.28	2.37	9	1	-38.16	419.61	16.1

N o t e : Gear ratio 0.5739; penetration depth 1 mm; incline angle 30°; hole radius 80 mm.

This is a primary condition in analysis of the serviceability of single-roller bits and the subsequent development of recommendations on improving their geometry.

Kinetic data of the bits are shown in Table 2 and Fig. 8. As one can see from the data of the table, the values of A' which define the relative wear of the cutters are not evenly distributed among the rows. The value of $A'_{max} = 419.61$ corresponds to a peak row, while $A'_{min} = 146.08$ corresponds to a heel row. This explains the uneven wear of the cutters of a single-roller bit with a spherical roller. In this case, peak rows will wear out more than 2.5 times faster than heel rows. The values of A''_{kl} defining the relative intensity of rock breaking in round strips of the bottom hole also differ. The value A''_{min} corresponds to the central section of the bottom hole, and the breaking conditions in the section are more difficult.

However, the criteria A'' for single-cone bits are greater than for tri-cone bits and are quite sufficient for effective rock breaking on a spherical bottom hole surface. Therefore, we shall be guided only by the criterion A'.

Calculation of the kinetic data for different arrangements and numbers of teeth of the cutters on a spherical roller revealed high specific contact work of the peak rows in each case.

The studies confirmed the need to find fundamentally new ways to improve the effectiveness of a single-roller bit.

The path of contact of a tooth with the rock at the bottom hole is a function of the height of the spherical strip on which it is working. Consequently, reducing the path of contact is possible by changing the height of the spherical strip on which the teeth of the row in question are working. This can be achieved by changing the form of the roller, making some of the rows in the form of concentric spheres of smaller radius. An effective bit roller geometry is found in this case by selecting the optimum combination of the number and arrangement of steps on the roller.

We shall examine the kinetic data of a bit (Table 3, Fig. 9), the six rows of which are arranged on a roller with three spherical strips of different curvature. The radius of each successive strip, starting at the edge, is less than the previous radius by 2.5 mm.

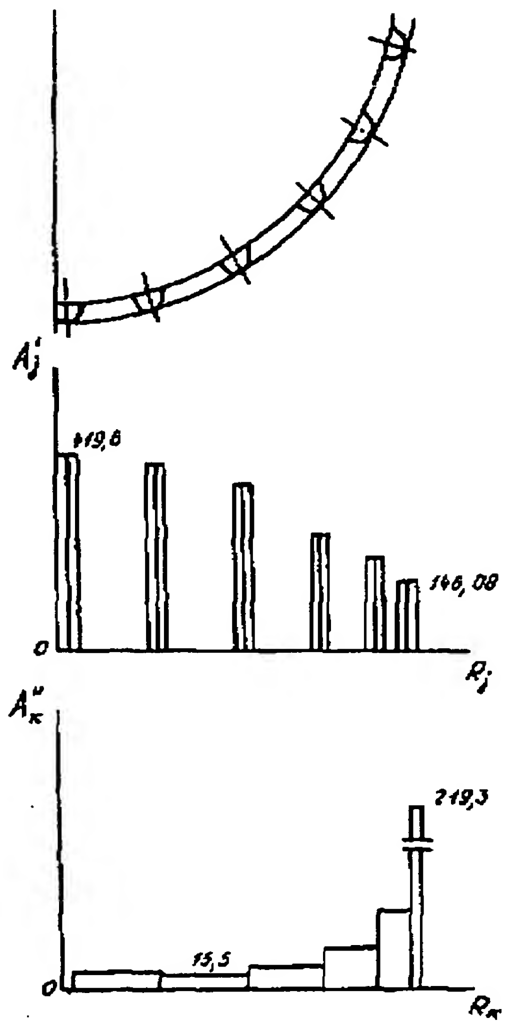

Fig. 8. Kinetic data for base bit I 161SZ-N.

Table 3

Geometric parameters					Kinetic characteristics		
Conven-tional row number J	Row radius Rj	Bottom hole radius Ro	Number of teeth in row	Row width D	Tooth speed v	Specific contact work A'	Specific volume work A''
1	79	77.5	9	1	11.33	70.83	119.8
2	80.4	69.5	18	1	0.61	92.95	15.5
3	75.5	55.5	9	1	-10.81	40.58	13.7
4	70.5	44.5	9	1	-18.63	74.38	9.2
5	52.5	19.0	9	1	-30.35	53.07	17.1
6	39.5	2.5	9	1	-36.59	86.40	33.3

N o t e : Gear ratio 0.65; penetration depth 1 mm; incline angle 30°; hole radius 80.5 mm; number of spheres 3.

As one can see from the kinetic data, the values of A' are more uniform by rows and much smaller than the value of A'_{max} than for a bit with a spherical roller.

It should be mentioned that the section in which the value of A'' is at a minimum is shifted toward the periphery of the bottom hole, where breaking is made easier by the graduated shape, which also makes it possible to increase the specific load on a tooth by reducing the number of teeth in constant contact with the bottom hole, and improves the clearing of cuttings from the bottom hole. The activity of the roller is also increased, as attested by the increase in the gear ratio from 0.57 to 0.86.

As a result, some reduction in the value of A'' does not cause a decrease in the mechanical rate of drilling. The probability of shearing of the cutters of rows that break a section of the bottom hole, to which the value of A''_{min} corresponds, obviously also does not increase.

The durability of a bit with a cone of graduated form increases, since the maximum intensity of the wear of cutters per revolution of the bit $A'_{max} \cdot i$ decreased in comparison to a bit with a spherical roller by a factor of more than three. Increasing the durability of the cutters of a bit will make it possible to improve its operational characteristics with sufficient bearing durability.

The study of the serviceability of single-roller bits, the comparative analysis of the causes of failure of bits and the nature of the distribution of specific contact and volume breaking work for rows and spherical strips of the bottom hole, respectively, and the solution of problems of the optimization of the geometry of drilling bits make it possible to draw the following conclusions:

1. The durability of single-roller bit I 161SZ-N with a spherical roller is limited by the abrasive durability of the peak rows.

2. For improving the durability of a single-roller bit, it is necessary to make the roller graduated in shape.

3. The most effective geometry for such a roller is a three-step geometry.

4. The use of a bit with a diameter of 161 mm with a roller of the recommended shape, other conditions being equal, increases durability and the mechanical drilling rate. In this process, the conditions for rock breaking and the removal of cuttings from the bottom hole are improved.

5. The effectiveness of the use of the bit is improved.

Tri-Cone Drilling Bit
III 215.9S-GNU (R45)

The tri-cone drilling bit fails as a result of wear (shearing) of the cone teeth or bearings, which, as a rule, is local in nature. Wear of the cone cutters and their bearings occurs on one of the rows or in one of the cones, respectively, in advance of others.

All this corresponds fully to the kinetic characteristics – the specific relative contact work of breaking A' and the specific relative volume work of breaking A'' or, more precisely, the ratios of their extreme values, which characterize the relative intensity of abrasive wear of the teeth of the rows of adjacent cones and the intensity of rock breaking in adjacent annular bottoms, respectively.

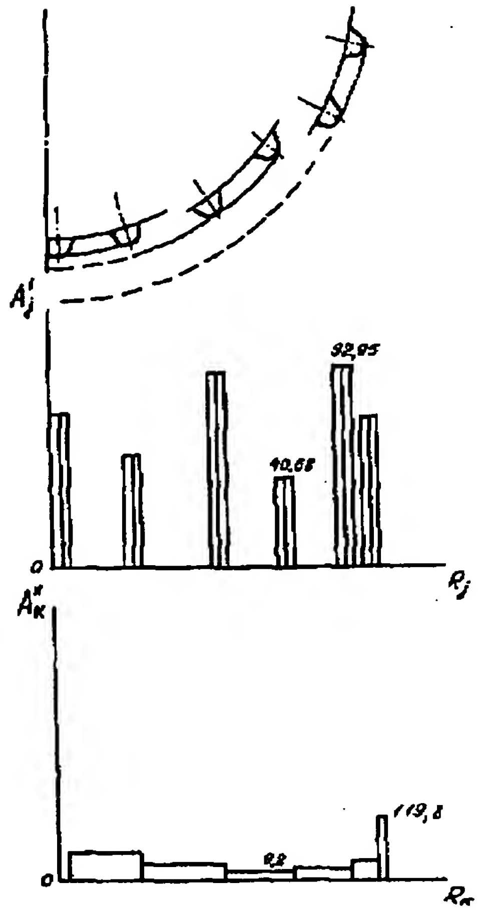

Fig. 9. Kinetic data of bit I 161SZ-N with graduated roller.

Based on the values of A'_{max}, one can predict with sufficient precision the row that wears ahead of others, and based on the values of A''_{min}, one can predict the row whose teeth have a higher probability of shearing and which determines the upper limit of the mechanical drilling rate, as well as the more heavily loaded bearing and, consequently, the bearing with leading wear.

These parameters are entered in the operational data certificate, which facilitates the development of recommendations on improving the bits, which can be done in three ways:

1. Creating equally loaded bearing of adjacent cones, i.e., balancing A''_{minI}, A''_{minII} and A''_{minIII}.

2. Increasing the overall (minimum) intensity of rock breaking, i.e., increasing A''_{min}.

3. Saving hard-alloy cutters, i.e., removing superfluous teeth in the rows, where A'' significantly exceeds A''_{min}.

Savings are possible in this process due to the increase in bit durability and the mechanical drilling rate and the decrease in bit cost. The optimum solution generally is possible with various combinations of these factors.

The development of recommendations is based on performing the following tasks.

1. Analysis of the kinetic data of the base bit design.

2. Determination of ways to improve the design.

3. Determination of the optimum combination of geometric parameters of the bit to produce the best kinematic performance data.

4. Comparative analysis of the serviceability of the recommended bit design and the base design in regard to kinetic data.

In addition, recommendations have been developed on the rational use of the bit design in question (the technological aspect). Recommendations are made for the base and improved designs (technical aspect). These recommendations have been developed in light of the possible replacement of the bit type or design.

The development of technological recommendations is also based on comparative analysis of the kinetic characteristics (certification of operating data) of drilling bits of various designs of the same or similar types. The difference lies only in the fact that the search for a more effective bit for the geological engineering conditions in question is conducted based on the bit catalog, i.e., among the existing range of bit designs, modifications and types.

The kinetic data for bit III 215.9S-GNU (R45) is presented in Table 4 and Fig. 10. The cutter design of this bit was not successful.

Table 4

Annular bottom hole sections from periphery to center			Cone I i = 1.494		Cone II i = 1.517		Cone III i = 1.516		Bottom breaking intensity
1	R, mm	D, mm	Z	A'	Z	A'	Z	A'	A"
1	108.0	1.0	51	31.85	48	31.84	57	31.85	732.02
2	106.6	2.9	31	3.85	48	3.74	57	3.73	87.92
3	104.2	1.8	17	3.71	16	3.69	19	3.69	29.37
4	102.5	1.8	17	3.67	16	3.71	19	3.72	29.89
5	100.2	2.7	-	-	16	3.81	19	3.82	21.23
6	97.5	2.7	-	-	16	4.04	19	4.05	23.10
7	94.8	2.7	-	-	16	4.35	19	4.37	25.81
8	87.8	2.7	15	4.97	-	-	-	-	13.53
9	85.1	2.7	15	5.27	-	-	-	-	14.80
10	82.4	2.7	15	5.60	-	-	-	-	16.22
11	79.7	2.7	15	5.94	-	-	-	-	17.80
12	75.2	2.5	-	-	-	-	14	6.73	19.92
13	72.7	2.5	-	-	-	-	14	6.71	20.57
14	70.2	2.5	-	-	-	-	14	6.70	21.27
15	67.5	2.5	-	-	-	-	14	6.68	22.02
16	65.1	2.6	-	-	13	6.68	-	-	21.25
17	62.5	2.6	-	-	13	6.74	-	-	22.31
18	59.9	2.6	-	-	13	6.80	-	-	23.47
19	57.4	2.6	-	-	13	6.86	-	-	24.75
20	53.0	2.7	11	6.65	-	-	-	-	22.01
21	50.3	2.6	11	6.65	-	-	-	-	23.15
22	47.7	2.6	11	6.65	-	-	-	-	24.43
23	45.0	2.6	11	6.65	-	-	-	-	25.87
24	40.4	3.0	-	-	-	-	9	7.25	25.70
25	37.4	3.0	-	-	-	-	9	7.31	27.98
26	34.5	3.0	-	-	-	-	9	7.39	30.69
27	30.4	3.0	-	-	6	7.68	-	-	24.13
28	27.5	3.0	-	-	6	7.80	-	-	27.14
29	24.5	2.9	-	-	6	7.95	-	-	30.95
30	16.8	2.3	5	8.48	-	-	-	-	40.23
31	14.5	2.3	5	8.75	-	-	-	-	48.15

32	12.2	2.2	5	9.09	-	-	-	-	59.25
33	10.0	2.1	5	9.51	-	-	-	-	75.50
34	7.9	2.1	2	9.94	-	-	-	-	39.97
35	6.1	1.5	2	10.48	-	-	-	-	55.07
36	5.0	0.8	2	10.62	-	-	-	-	67.63
37	4.9	0.6	2	11.61	-	-	-	-	74.62

As a result of the uneven distribution of minimum intensities of rock breaking on rows of adjacent cones, bearing overload of cone *I* will be observed.

The same cone also determines the comparatively low potential mechanical drilling rate, i.e., the value $A''_{min} = 13.526$ for type S bits with a diameter of 215.9 mm is insufficient. The following adjustment of the cutter arrangement is necessary for improving the potential effectiveness of this bit.

1. Increase the number of teeth in the second heel row of cone *I* from 15 to 20.

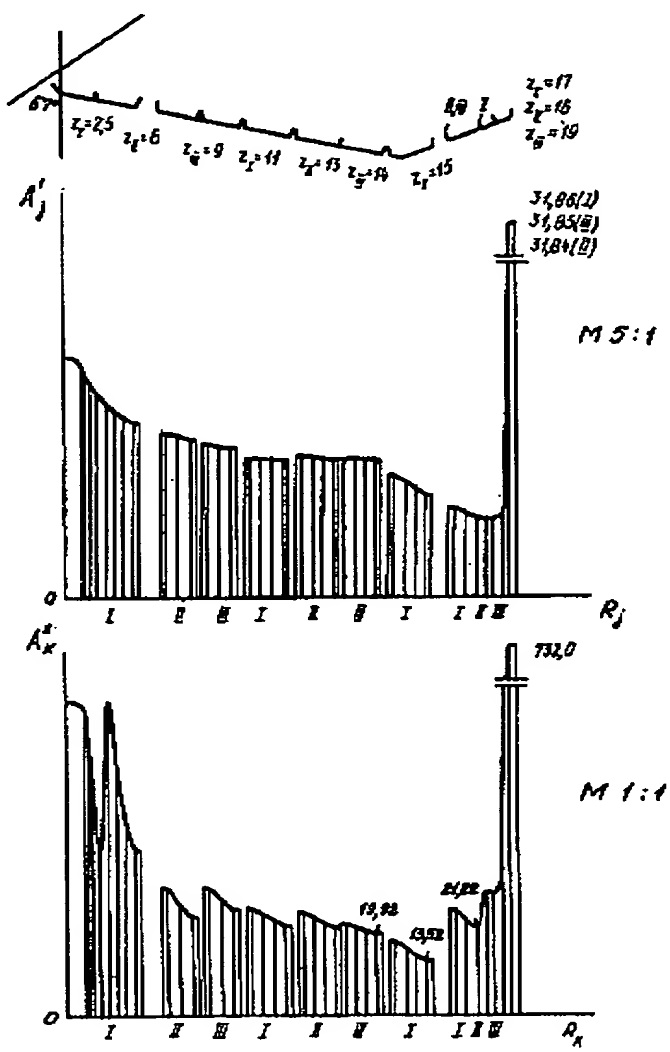

Fig. 10. Kinetic data of base bit III 215.9S-GNU (R45).

2. Reduce the number of teeth in the heel row of cone *II* from 16 to 14.
3. Reduce the number of teeth in the heel row of cone *III* from 19 to 16.
4. Increase the number of teeth in the second heel row of cone *III* from 14 to 15.

The kinetic data for the recommended arrangement of cone cutters is shown in Table 5 and Fig. 11. The advantages of the recommended cutter arrangement alternative.

Table 5

Annular bottom hole sections from periphery to center			Row wear intensity						Bottom breaking intensity A"
1	R, mm	D, mm	Cone I i = 1.494		Cone II i = 1.517		Cone III i = 1.516		
			Z	A'	Z	A'	Z	A'	
1	108.0	1.0	51	32.04	42	31.84	48	31.64	662.95
2	106.6	2.9	51	4.12	42	3.74	48	3.80	82.08
3	104.2	1.8	17	3.88	14	3.69	16	3.69	25.98
4	102.5	1.8	17	3.75	14	3.71	16	3.67	27.09
5	100.2	2.7	-	-	14	3.80	16	3.71	17.88
6	97.5	2.7	-	-	14	4.02	16	3.67	19.29
7	94.8	2.7	-	-	14	4.32	15	4.14	21.28
8	87.8	2.7	20	4.50	-	-	-	-	15.33
9	85.1	2.7	20	4.78	-	-	-	-	17.88
10	82.4	2.7	20	5.08	-	-	-	-	19.63
11	79.7	2.7	20	5.41	-	-	-	-	21.60
12	75.2	2.5	-	-	-	-	15	6.40	20.31
13	72.7	2.5	-	-	-	-	15	6.39	20.98
14	70.2	2.5	-	-	-	-	15	6.38	21.71
15	67.6	2.5	-	-	-	-	15	6.37	22.50
16	65.1	2.6	-	-	13	6.65	-	-	21.15
17	62.5	2.6	-	-	13	6.71	-	-	22.20
18	59.9	2.6	-	-	13	6.77	-	-	23.36
19	57.4	2.6	-	-	13	6.83	-	-	24.63
20	53.0	2.7	11	6.19	-	-	-	-	20.45
21	50.3	2.6	11	6.20	-	-	-	-	21.56
22	47.7	2.6	11	6.21	-	-	-	-	22.80
23	45.0	2.6	11	6.22	-	-	-	-	24.20
24	40.4	3.0	-	-	-	-	9	7.00	24.82
25	37.4	3.0	-	-	-	-	9	7.07	27.05
26	34.5	3.0	-	-	-	-	9	7.15	29.72
27	30.4	3.0	-	-	6	7.66	-	-	24.05
28	27.5	3.0	-	-	6	7.78	-	-	27.06
29	24.5	2.9	-	-	5	7.92	-	-	30.87
30	16.8	2.3	5	8.17	-	-	-	-	38.78
31	14.5	2.3	5	8.45	-	-	-	-	46.55
32	12.2	2.2	5	8.82	-	-	-	-	57.47
33	10.0	2.1	5	9.26	-	-	-	-	73.48
34	7.9	2.1	2	9.71	-	-	-	-	39.04
35	6.1	1.5	2	10.28	-	-	-	-	54.02
36	5.0	0.8	2	10.53	-	-	-	-	67.09
37	4.9	0.6	2	11.35	-	-	-	-	73.57

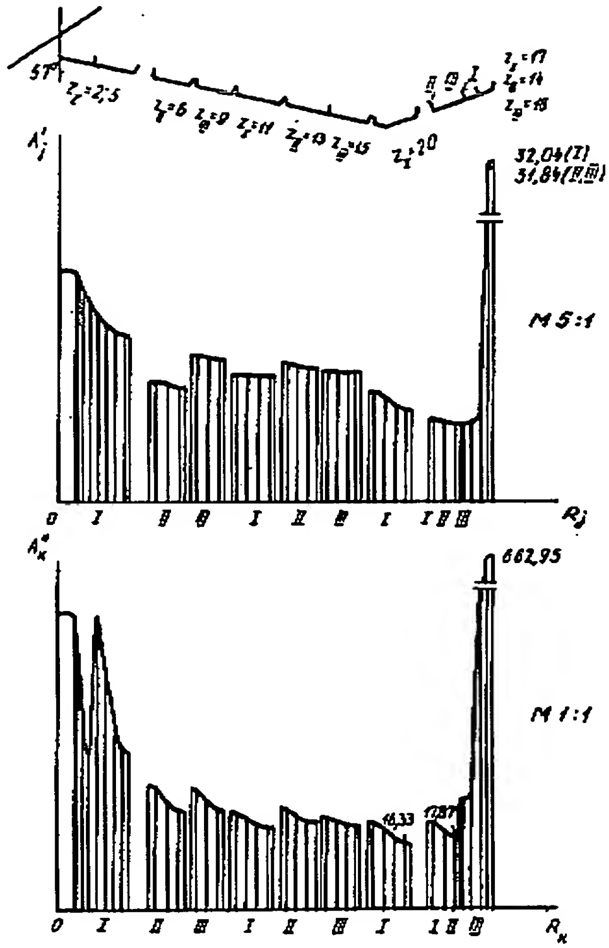

Fig. 11. Kinetic data of bit III 215.9S-GNU
(R45) with modified arrangement of cone cutters

1. The mechanical rate potential increases by 20% as a result of the increase in A''_{min} from 13.526 to 16.328.

2. The durability of the bit increases as a result of the more even distribution of loads among bearings of adjacent cones: $A''_{min\mathrm{I}} = 16.328$; $A''_{min\mathrm{II}} = A''_{min\mathrm{III}} = 17.877$.

The bit is first selected according to its classification code and then is refined based on comparative analysis of the potentials of bits of the same or similar types. This selection adjustment is made in light of the actual nature of wear of the cutters and bearings of adjacent cones.

The potential of a bit is determined according to the kinetic data. For fast solution of the problems stated, the bit potential is entered on special cards (Tables 6 and 7, second column). Table 6 is for the base model; Table 7 is for the recommended and perfected design.

If drilling was being conducted with a bit with a lesser potential, it is replaced with the type of bit in question (see Tables 6 and 7, first column). If drilling was being conducted with this type of bit, and its productivity is insufficient, it is replaced with another bit of the same type or a similar type with greater potential (see Tables 6 and 7, third column).

Table 6

Cases of replacement with bit III 215.9S-GNU (R45	Potential of bit III 215.9S-GNU (R45)	Cases of replacement of bit III 215.9S-GNU (R45)
1. Cutters worn; $A'_{max} > 31.84$	$A'_{max} = 31.84$ $A'_{min} = A'_{minI} = 13.526$	1. Leading wear of cutters; replaced with bit with $A'_{max} < 31.84$
2. Shearing of teeth; $A''_{min} < 13.526$		2. Shearing of cutters; replaced with bit with $A''_{min} > 13.526$
3. Leading wear of bearings; $A''_{minII,\,III} - A''_{minI} > 7.699$	$A''_{minII} = A'_{minIII} = 21.225$	3. Leading wear of bearings; replaced with bit with $A''_{minII,\,III} - A''_{minI} < 7.699$
4. Low mechanical drilling rate; $A''_{min} < 13.526$		4. Low drilling rate; replaced with bit with $A''_{min} > 13.526$

Table 7

Cases of replacement with bit III 215.9S-GNU (R45	Potential of bit III 215.9S-GNU (R45)	Cases of replacement of bit III 215.9S-GNU (R45)
1. Cutters worn; $A'_{max} > 32.04$	$A'_{max} = 32.04$ $A'_{min} = A''_{minI} = 16.328$	1. Leading wear of cutters; replaced with bit with $A'_{max} < 32.04$
2. Shearing of teeth; $A''_{min} < 15.328$		2. Shearing of cutters; replaced with bit with $A''_{min} > 16.328$
3. Leading wear of bearings; $A''_{minII,\,III} - A''_{minI} > 1.549$	$A''_{minII,\,III} = 17.877$	3. Leading wear of bearings; replaced with bit with $A''_{minII,\,III} - A''_{minI} < 7.699$
4. Low mechanical drilling rate; $A''_{min} < 16.328$		4. Low drilling rate; replaced with bit with $A''_{min} > 16.328$

Drilling Head KS 212.7/60TKZ

For more detailed analysis of the serviceability of drilling head cone cutters, kinetic characteristics of cones *I* and *II* (Table 8) and *II* and *IV* (Table 9) were computed.

Figure 12 *a* shows that the intensity of abrasive wear of the nose teeth is 10 times higher, even in comparison to the teeth of heel rows. Such a value of A'_j is not encountered in computing the kinetic characteristics of tri-cone bits. Consequently, leading wear of nose teeth is absolute, to a significant degree.

Figure 12 *b* clearly shows at once several shortcomings of the design of this drilling head.

1. All Three minimums of the function $A''_k = A\ (R_k)$ fall in annular bottom holes covered by cutters of cones *I* and *III*. Therefore, the bearing overload of this pair of cones is obvious, which, in the final analysis, results in leading wear.

2. The least value of A''_k for cones I and III falls in the pre-core region. This significantly reduces the durability of the cone bearings due to constant resting of this pair of cones on the edges. It must be mentioned that the durability of the edge teeth under this kind of loading will be reduced significantly.

3. The gear ratios of cones *I* and *II* and cones *III* and *IV*, respectively, are equal and amount to $i_{I,III} = 1.838$ and $i_{II,IV} = 1.891$; i.e., they are near 2. This indicates that the edge teeth, especially of cones I and III, will come into contact with the rock twice in a single revolution of the drilling head, i.e., at diametrically opposite points of the core. In this process, no matter how these points of the cone pair in question are shifted, the formation of blocks in the pre-core annular bottom hole and the breaking up of the blocks by the cones are clear. And this is the main shortcoming of the design of drilling head KS 212.7/60TIKZ, since the kinetics of the teeth in question is not doubled.

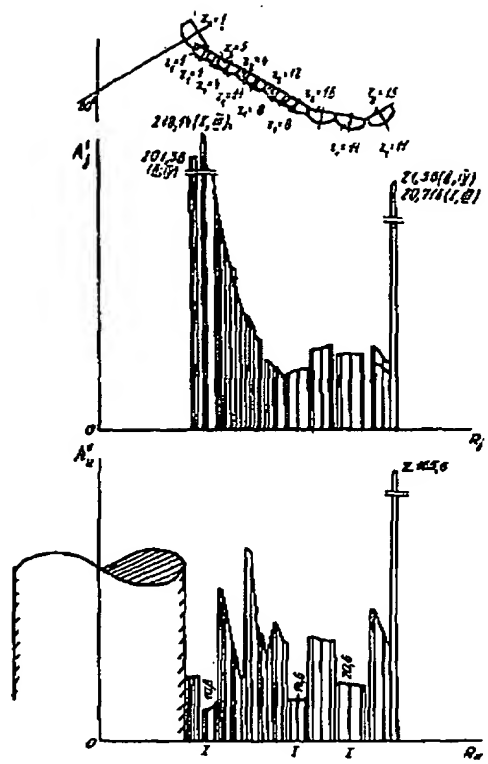

Fig. 12. Kinetic data of base drilling head KS 212.7/60TKZ.

Table 8

Geometric parameters					Kinetic characteristics		
Conventional row number I	Row radius R_I, mm	Number of teeth in conventional row Z	Bottom hole radius R_o, mm	Tooth length of conventional row D, mm	Tooth speed v, m/sec	Specific contact work *A'*	Specific volume work *A''*
1	58.0	11	105.9	1.0	-0.37	20.716	68.4
2	58.0	11	104.4	2.9	-1.19	3.781	12.8
3	58.0	11	101.5	2.9	-2.77	4.281	14.8
4	58.0	11	98.7	2.9	-4.29	4.948	17.6
5	55.8	11	93.7	3.2	-4.81	2.252	19.6
6	53.6	11	89.8	3.2	-4.73	5.260	20.6
7	51.5	11	86.6	3.2	-4.38	5.116	20.6
8	42.3	8	73.5	2.7	-2.30	4.245	14.8
9	40.4	8	70.9	2.7	-1.82	4.060	14.6
10	38.5	8	68.2	2.7	-1.39	3.914	14.6
11	35.8	8	65.6	2.6	-0.10	3.653	14.2
12	32.5	8	63.0	2.6	1.78	4.298	17.4
13	26.8	11	56.9	3.5	4.16	5.240	38.4
14	23.5	11	53.3	3.5	5.50	7.615	51.4
15	18.3	4	47.7	3.0	7.65	11.194	29.8
16	15.8	4	44.7	3.0	8.52	13.109	37.4
17	12.5	1	41.2	4	9.92	15.756	13.0
18	9.0	1	37.8	2.7	11.57	218.148	10.8

N o t e : Gear ratio 1.838; penetration depth 1 mm; incline angle 1.047°; hole radius 106.35 mm.

Table 9

Geometric parameters					Kinetic characteristics		
Conventional row number I	Row radius R_I, mm	Number of teeth in conventional row Z	Bottom hole radius R_o, mm	Tooth length of conventional row D, mm	Tooth speed v, m/sec	Specific contact work A'	Specific volume work A''
1	58.0	15	105.9	1.0	-2.01	21.563	97.2
2	58.0	15	104.4	2.9	-2.80	4.418	20.2
3	58.0	15	101.5	2.9	-4.34	5.112	24.0
4	58.0	15	98.7	2.9	-5.82	5.903	28.6
5	48.9	16	82.7	3.3	-5.18	5.782	35.6
6	46.8	16	79.4	3.3	-4.82	5.640	36.2
7	44.8	16	76.1	3.3	-4.57	6.644	37.2
8	35.0	12	66.2	3.0	0.00	3.758	21.6
9	32.2	12	63.1	3.0	1.16	4.108	24.8
10	29.4	12	60.1	3.0	2.37	4.865	31.0
11	24.0	4	53.9	3.6	4.50	6.974	16.4
12	20.7	4	50.3	3.6	6.89	8.881	22.4
13	17.4	5	46.8	3.4	7.34	11.377	38.6
14	14.1	5	43.4	3.4	8.85	14.696	53.8
15	3.8	2	33.7	2.2	14.02	201.385	22.4

Note: Gear ratio 1.891; penetration depth 1 mm; incline angle 1.047°; hole radius 106.35 mm.

Based on the information presented, one can conclude that the design of drilling head KS 212.7/60TKZ has low serviceability potential of the cutters.

To enhance the potential of this design, it is necessary to change the kinetics of the drilling head as follows:

reduce A'_{max} of the function $A'_j = A(R_j)$;

set A''_{min} equal, if not for both pairs of cones, then for at least one pair (in this case, cones *I* and *III*);

double the kinetics of the edge teeth of the cones where possible.

Of course, the effect will be at a maximum if all three conditions are fulfilled. However, this is an ideal case. One can fulfill one of the conditions while leaving the others unchanged or even worse.

To be specific, one such solution is shown by its kinetic characteristics (Table 10 and 11 and Fig. 13).

Table 10

Geometric parameters					Kinetic characteristics		
Conventional row number I	Row radius R_I, mm	Number of teeth in conventional row Z	Bottom hole radius R_o, mm	Tooth length of conventional row D, mm	Tooth speed v, m/sec	Specific contact work A'	Specific volume work A''
1	58.0	12	105.9	1.0	-4.31	24.357	87.8
2	58.0	12	104.4	2.9	-5.04	5.707	20.8
3	58.0	12	101.6	2.9	-6.50	6.563	24.6
4	58.0	12	98.7	2.9	-7.96	7.491	29.0
5	55.8	12	93.1	3.2	-8.54	7.980	32.8
6	53.7	12	89.8	3.2	-8.09	7.773	33.0

7	51.5	12	86.6	3.2	-7.63	7.566	33.4
8	42.3	8	73.2	3.6	-5.15	6.254	21.8
9	39.8	8	69.5	3.6	-4.49	5.895	21.6
10	37.0	8	66.4	2.6	-3.31	5.205	20.0
11	34.0	8	63.8	2.6	-1.62	4.324	17.2
12	27.4	6	57.1	3.4	1.60	4.596	15.4
13	24.1	6	53.8	3.4	3.15	5.954	21.2
14	19.5	4	48.7	3.2	5.20	8.641	22.6
15	16.5	4	45.5	3.2	6.55	11.0554	31.0
16	11.8	3	41.7	2.8	9.39	17.619	40.4
17	8.3	3	38.8	2.8	11.45	22.390	33.4

Note: Gear ratio 1.972; penetration depth 1 mm; incline angle 1.047°; hole radius 106.038 mm.

Table 11

Geometric parameters					Kinetic characteristics		
Conventional row number I	Row radius R_I, mm	Number of teeth in conventional row Z	Bottom hole radius R_o, mm	Tooth length of conventional row D, mm	Tooth speed v, m/sec	Specific contact work A'	Specific volume work A''
1	58.0	16	105.9	1.0	-6.08	26.572	127.8
2	58.0	16	104.4	2.9	-6.79	6.966	34.0
3	58.0	16	101.6	2.9	-8.20	7.900	39.6
4	58.0	16	98.7	2.9	-9.62	8.884	45.8
5	49.0	16	82.7	3.3	-8.41	8.547	52.6
6	46.8	16	79.4	3.3	-7.92	8.301	53.2
7	44.8	16	76.1	3.3	-7.43	8.052	53.8
8	35.6	12	66.6	2.9	-2.94	5.180	29.8
9	32.8	12	63.6	2.9	-1.55	4.448	26.8
10	29.9	12	60.7	2.9	-0.14	4.052	25.4
11	24.4	5	54.2	3.6	2.21	5.307	15.6
12	21.2	5	50.7	3.6	3.71	6.982	22.0
13	16.6	3	46.0	3.1	5.91	10.493	21.8
14	13.9	3	42.8	3.1	7.13	13.239	29.6
15	9.5	3	39.4	3.9	9.81	229.378	45.0

Note: Gear ratio 2.039; penetration depth 1 mm; incline angle 1.047°; hole radius 106.038 mm.

Positive aspects of the design of drilling head KS 212.7/60TKZ:

1. The minimum of A'' was increased from 10.8 to 15.4. Consequently, the potential mechanical drilling rate was increased.

2. The minimum of A'' was transferred from the pre-core region by approximately half the annular bottom hole. Consequently, the bending moment on the bearings of cones *I* and *III* was reduced.

3. The minimum $A''_{min} = 15.4$ itself covers a comparatively narrow bottom hole ring and only slightly affects the uneven loading of the bearings of cone pairs *I-III* and *II-IV*, i.e., the minimum of successive values are approximately the same.

4. The kinematics of the drilling head and, consequently, the kinematics of the edge teeth are repeatedly doubled. Consequently, core forming will be performed steadily, without the formation of blocks in the pre-core bottom hole region, regardless of gear ratios.

The negative factor in the design is an increase in the value of A'_{max} from 218.14 to 226.39, which generally results in heavy abrasive wear of the edge teeth. However, this was done

consciously in the search for a more effective drilling head design. We shall mention that in the base model, $A'_{max} = 218.14$ for only two teeth of cones *II* and *IV*. In the proposed design, this region is doubled with a value $A'_{max} = 226.39$ for 14 teeth. Consequently, this characteristic does not impact the durability of the cutters of the drilling head as a whole.

Hence, based on an analysis of the kinetics of the cutters of drilling head KS 212.7/60TKA, an effective modification has been found.

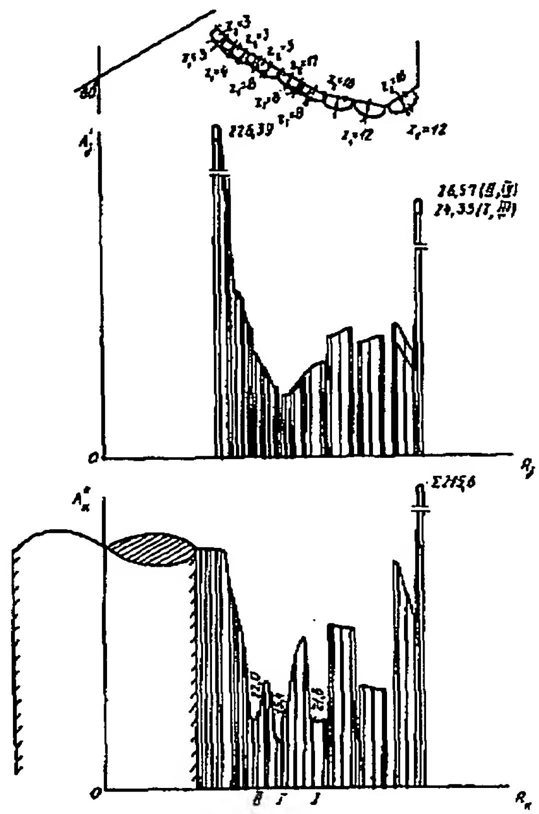

Fig. 13. Kinetic data for drilling head KS 212.7/60TKZ with modified arrangement of cone cutters.

This optimization solution does not exhaust the potential of this drilling head design. However, it has been expedient to discuss it as a first approximation, since it would otherwise be necessary to make adjustments to the design of the cone bearings.

CONCLUSION

The operation of a rock-breaking drilling tool, as the studies have demonstrated, is the result of the operation of quite a complex dynamic system. And this system is self-governing to some degree.

It is important to note that despite the complexity of this system, ways have been found to construct a determinate mathematical model of its operation. Without question, this result was facilitated to a great extent by theoretical research previously performed by other authors. This pertains to all the basic issues of the structure of the model: construction of parametric equations of the trajectories of movement of the cone cutters, analytical determination of their gear ratios, and the structure of kinetic criteria of the serviceability of drilling bits.

The pattern discovered in the power expenditure from forces of resistance to movement in rotary movement unquestionably makes a substantial contribution to the conception of the bottom hole dynamics of drilling bits. By transferring the properties of this pattern to a mechanical model of a drilling bit, one can easily explain the radial movements of the bit at the bottom hole. An important issue here is the fact that the mode of rotation of the drilling bit around a single axis (the hole axis) is not effective in light of the principle of the lowest power expenditure. The existence of a large number of rotation modes with lower power consumption, but with rotation around two parallel axes, is the basic cause of radiation oscillations of the bit.

However, the drilling bit as a dynamic system during operation cannot in itself enter another rotation mode different from the mode of rotation around a single axis of the bit. It is here that the role of asymmetry of the action of reactive forces from the bottom hole is obvious. Such asymmetry, as a rule, is a result of anisotropy of the rock, asymmetry of the cone suspension and intensity of the breaking of rock in adjacent drilled annular bottom holes. These factors promote entry of the bit into a mode of rotation with the lowest power expenditure – a mode of rotation around two parallel axes. We note that they only promote this result inasmuch as, in a mode of rotation of the bit around a single axis, these factors and properties of the pattern of power expenditure (6) do not hinder each other in achieving the goal – radial movement of the bit at the bottom hole. When the goal is achieved, these factors and properties of the pattern (6) instantly come into conflict in the aspect of stabilization of the bit in relation to the hole axis. Such are the properties of the pattern obtained, and it is in just this regard that the physics of vibration processes of rock-breaking drilling tools at the bottom hole should be considered.

The conflict of the very properties of the pattern of power consumption in various rotation modes generally accomplishes functions of stabilization of the drilling bit with a minimum possible range of eccentricity.

As the studies have demonstrated, such a range of variation of eccentricity in the operation of roller cone drilling bits does not have a large impact on calculations of the kinetic characteristics by formula (5). The coordinates of their extreme values remain constant in this process. This factor is especially important in analyzing the serviceability of drilling bits. Otherwise, it would be necessary to make the mathematical model of their operation more complex.

For the purpose of deeper study and analysis of the kinetics of cone cutters, such an improvement in the model of bit operation will probably be necessary all the same. Even now one must take this into consideration in qualitative analysis of the mechanism of interaction of cone cutters with the bottom hole surface, and one must keep in mind the exclusion of additional factors that destabilize the drilling bit at the bottom hole. Such factors include, for example, the introduction of asymmetry into the arrangement of cutters of the rows by placement of the teeth with different geometric spacing. A mechanism with paired teeth can serve as an example.

In general, it is always necessary to keep in mind that a change in any geometric parameter of a drilling bit has an effect on its dynamic equilibrium.

The effect of the angle between the cone and bit axes on the dynamic equilibrium is demonstrated quite convincingly in the example of spherical rollers. In this case, the nature of the destabilization of rollers on a spherical bottom hole is explained in the light of all force factors.

The dynamic instability of drilling bits at the bottom hole absolutely determines the need for introducing stabilizing elements into this dynamic system in each individual case. The example of the three-roller stabilizer which is given confirms this concept.

It must be mentioned that the bottom hole operation of a drilling bit is a complex dynamic system. It does not appear to be possible to express all its aspects analytically in a single complex; the fewer restrictions imposed, the more effective the solution. For example, removing just one restriction – the absence of deformability of the bottom hole surface – made it possible to construct the analytical structure of kinetic criteria for the assessment of the serviceability of drilling bits which provide an objective picture of the process of rock breaking in drilling.

Hence the effectiveness of feedback is in the solution of optimization problems in the process of improving drilling tools.

A whole set of such relationships has been worked out on this basis. The optimization problem, even at this level, is not limited to these relationships.

The outward appearance of a drilling bit of any size can be reduced to a near ideal form; i.e., all the bit design alternatives for the entire set of geometric parameters can be tested. At this stage, the search for solutions and ways of implementing them is conducted within the strictest framework of restrictions based on the actual functional dependence of kinetic criteria of particular geometric parameters. For example, analysis of the kinetics of single-roller bit I 161SZ-N demonstrated that the failure of the bit observed in practical work is conditioned by the comparatively greater specific contact work of breaking, as a result of which leading wear of the teeth of nose rows occurs. However, this is ineffective due to the comparatively low durability of the bit and the sharp drop in the mechanical drilling rate as a result of intensive wear of the cutting edges of the teeth of the nose rows. In the final analysis, there is an adverse impact on the footage drilled with the bit.

The specific contact work of breaking is in direct proportion to the depth of embedding of the rows in the spherical bottom hole surface. This was the starting point in the search for an effective solution. The embedding depths in this case can be differentiated by changing the form of the spherical roller and giving it a graduated shape.

In this way, it was possible to bring the specific contact work of breaking for teeth of the nose rows and heel rows into line, to improve the conditions of removal of cuttings from the bottom hole, to increase the specific load on the teeth, and to move the minimum value of the specific volume work of breaking away from the central area of the bottom hole. The result is an increase in the value of the bit, the mechanical drilling rate and the dynamic stability of the bit in relation to the hole axis due to the high intensity of rock breaking in the central area of the bottom hole. This, in turn, significantly improves the probability of drilling without deviation.

Such a solution clearly guarantees improvement in the effectiveness of a drilling bit during operation, other conditions being equal.

An entirely different approach to improving tri-cone bits was demonstrated in the example of bit III 215.9S-GNU (R45). This bit fails due to leading wear of the bearings. However, as already noted, the durability of the bearings in tri-cone bits is determined by catastrophic wear of one of the bearings due to overloading. This is entirely consistent with the ratios of minimum values of specific volume work of breaking of the rows of adjacent cones.

The statement and solution of the optimization problem arise from this. The problem consists of balancing the minimum values of specific volume work of breaking, by adjusting them to one of the values, for example. This can be achieved comparatively easily by varying the number of teeth in the rows in question. Such an approach makes it possible to effect a guaranteed increase in the potential of tri-cone bits in regard to the criterion of durability at the bottom hole.

The optimization problem demonstrated in the example of drilling head KS 212.7/60TKZ is quite different in nature. The usage experience of these drilling heads has demonstrated that their durability at the bottom hole is limited by the durability of the cutters in edge areas. Analysis of the kinetics of the cutters of these cones, in turn, demonstrated that the adjacent pre-core annular sections of the bottom hole are covered by individual, unpaired teeth of the rows of adjacent cones with comparatively great specific contact work of breaking, with local rock breaking zones which are not distributed over the entire annular bottom hole. The conformity of the cause of failure of the drilling head to the kinetics of the cutters identified in this way made it possible to find an effective solution. It is easy to ascertain that all the optimization solutions presented are based on this kind of methodology. This, in turn, ensures a guaranteed increase in the potential capabilities of rock-breaking drilling tools, improved on a determinate basis.

REFERENCES

1. Bilanenko, N. A., B. L. Steklyanov, A. A. Torgashov and S. P. Batalov. "Testing of an Experimental Lot of Bits Developed Using an Analytical Model." *RNTS*, ser. "Bureniye." Moscow: VNIIOENG, 1983, issue 5.

2. Golubintsev, O. N. *Mechanical and Abrasive Properties of Rocks and Their Drillability*. Moscow: Nedra, 1968.

3. Yegerev, A. G. *Theoretical Bases for the Design and Use of Bits*. Moscow: Gostoptekhizdat, 1945.

4. Kapitsa, P. L. "Stability and Passage Through Critical Revolutions of Rapidly Rotating Rotors in the Presence of Friction." *Zhurnal tekhnicheskoy fiziki*, v. IX, issue 2, 1939.

5. Kalinin, A. G. "Mechanism of the Formation of Bores of Exploratory Wells." *Inf. Soobshch.*, ser. "Tekhnika I tekhnologiya geologorazvedochnykh rabot I organizatsiya proizvodstva." Moscow: VIEMS, 1968, No. 27.

6. Kudekov, Yu. F. "Design Features on Small Three-Roller Expander-Stabilizers." In collected proceedings of SAIGIMS, "Improvement of Combined Drilling of Exploratory Wells for Solid Minerals with Diamond and Hard Alloy Crowns and Roller Cone Bits." Tashkent, 1974, pp. 49-52.

7. Golyakov, V. S. "Some issues of the Mechanics of the Operations of Roller Cone Bits." *Neftyanoye khozyaystvo*, 1957, No. 9, pp. 18-25.

8. Simonov, V. V., and V. G. Vyskrebtsov. *Operation of Roller Cone Bits and Improving Them*. Moscow: Nedra, 1975.

9. Steklyanov, B. L. "Classifier for the Mechanism of Interaction of Roller Cone Rock-Breaking Tools with a Bottom Hole Surface." Tashkent: Izv. AN UzSSR, ser. "Tekhnicheskiye nauki," 1962, No. 3.

10. Steklyanov, B. L., and I. I. Shamansurov. "Procedure for Determining the Position of the Momentary Rotation Axis of Roller Cones and Assessment of the Effectiveness of the Bit."

In collection "*Improvement of the Engineering and Technology of Exploratory Rock-Drilling Work.*" Tashkent, 1972, pp. 52-61.

11. Steklyanov, B. L., and I. I. Shamansurov. "General Kinematic Equation of the Movement of Cone Teeth." Tashkent: DAN UzSSR, 1971, No. 2.

12. Steklyanov, B. L. "On the Multifaceted Forming of Hole Cross Sections." Tashkent: Izv. AN UzSSR, ser. "Tekhnikcheskiye nauki," 1965, No. 1.

13. Steklyanov, B. L. "Kinetic Data of a Roller Cone Bit." *Gazovaya promyshlennost*, 1980, No. 11.

14. Steklyanov, B. L., N. A. Bilanenko and K. G. Valiyeva. "Procedure for Optimization of the Designs of Drilling Bits and Selection of Bit Types and Optimization Conditions under Predetermined Geological Engineering Conditions." Informational pamphlet on Advanced Production Experience. Tashkent: UzNIINTI, 1987.

15. Steklyanov, B. L., N. A. Bilanenko, A. I. Ilkovskiy and A. G. Kolugar. "Procedure for Solving Optimization Problems in the Selection and Optimization Drilling Bits." *RNTS*, ser. "Bureniye." Moscow: VNIIOENG, 1982, issue 12.

16. Shamansurov, I. I., and B. L. Steklyanov. *Kinematics of Roller Cone Bits.* Tashkent: FAN, 1977.